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EFFECT OF VALVE PORT GAS INERTIA ON VALVE DYNAMICS-PART II: FLOW RETARDATION AT VALVE OPENING

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ABSTRACT

A mathematical model of non-linear reciprocating air-compressor discharge valve system unsteady response was presented in Part I. As a first step, to demonstrate the application of the unsteady modeling approach, a comparison of discharge disc-valve response was made with a more popular quasi-steady modeling technique. For unsteady disc-valve response, contraction coefficients in the vicinity of the valve and orifice sections were determined from static measurements of discharge and lift-force coefficients. Under the assumptions that the contraction coefficients do not change appreciably from the static flow pattern during the dynamic response of the disc-valve, it is shown by comparing results of both modeling techniques that the inertia of the fluid, which initially retards the flow between the disc-valve and seat, has a dominant effect on valve response for suction to pressure ratios commonly used in most air compressor designs. Lastly some calculated time histories are shown which illustrate the effects of mechanical disc-valve systems natural frequency, mass and stiffness, damping and valve diameter on unsteady flow and valve response.

INTRODUCTION

The problem of reciprocating air compressor valve response is recognized as one of considerable importance and is a factor in air-compressor design. In a typical air compressor the suction and discharge valve consists usually of stainless steel beams or rings of high tensile strength which are riveted to a valve assembly block. During the compression and expansion process the air is forced through a set of ports located in the valve assembly block and directed against the valve element. A number of porting configurations are generally employed which depend on the manufacturers' specification of the compressor design. During the suction cycle one side of the valve system remains exposed to the atmosphere where the air is directed through the porting arrangement. Unlike the suction valve configuration, the discharge porting configuration may be different and the valve is usually enclosed by a discharge cavity of some finite volume into which the discharge flow emerges.

The modeling of the dynamic behavior of a reciprocating air-compressor valve has received much attention by many investigators, REF (1,2). It is well established that the flow between the valve and seat has a pronounced influence on the dynamic characteristics of the valve. Such a flow is very complex and its type changes extremely with the lift force acting on the valve. Mean pressure differences are created across the valve system during which the fluid stream and the motion of the valve element may interact causing the valve element to oscillate at a frequency approximately equal to that of the natural frequency of the mechanical valve system. This,in some instances, gives rise to increased pressure and flow oscillation superimposed on the mean pressure and flow. These oscillations depend on the amount of feedback energy from the oscillating valve plate and the performance of the air compressor is affected by its presence.

One of the most popular approaches, REF (1), to the mathematical modeling of the flow and the valve dynamics in a reciprocating air compressor is the use of steady state experimental flow and valve force area data. This approach has shown to give reasonable prediction of performance and discharge valve motion. In using this modeling technique some basic assumptions are applied. Of these the dynamics of the fluid stream is separated from the dynamics of the mechanical valve element by neglecting the influence of the velocity of the valve element since the mechanical valve system's natural frequency is considerably higher than that of the pumping frequency of the air. The most critical assumption eliminates the unsteady effect of the air in the restriction passages of the valve system by arguing that the mass of air in such passages are restricted in volume, hence having little effect on the incoming or discharging flow. However, in most recent applications of this approach to the modeling of the dynamic behavior of discharge and suction valves REF (3,4,5) a comparison with measured suction valve motion makes one suspect that it is important to include the inertia of the air in the valve restriction passage in certain cases. This inertia is associated with the flow stream which causes a retardation in valve motion during the valve's initial open time period. As the valve "momentarily hovers" over the orifice-valve seat, large positive pressures are developed in the orifice section. The valve lift force begins to increase immediately afterwards, causing the valve to accelerate from its seat. The inertia of air, in this context, does not represent an "effective/apparent gas inertia" as commonly used in most vibration modeling schemes.

In the following results of the quasi steady and the unsteady modeling techniques were compared and are presented for two discharge to suction pressure ratios. Computed valve displacements indicate that the inertia of the air in the disc valve passage and orifice region seems to be most pronounced for low discharge to suction pressure ratios. This discharge valve system seems to bear some similarity to a suction valve since for low discharge to suction pressure ratio it remains opened for a half cycle of reciprocating piston motion. Several parametric investigations were also conducted. The first by varying the stiffness of the mechanical spring. The second by varying the stiffness of the mechanical spring and the mass of the disc valve simultaneously maintaining the same natural frequency. The final investigations considered the influence of mechanical damping and the response of a large diameter disc valve. Although there is no experimental verification of the results predicted for the discharge disc valve system, they exhibit the same trends as experimental results found by various investigators, REF (3,4,5), for similar valves.

DISC-VALVE ENTRANCE AND EXIT CONTRACTION COEFFICIENTS

The flow between a disc valve and seat has a pronounced influence on the dynamic response of the disc valve system. In particular, various classes of flow may exist depending on the position of the disc valve and its type may change extremely with the lift force. For small distances between the valve and its seat the flow is laminar, and viscous forces dominate the magnitude of valve lift force. The range of the laminar flow region depends upon the diameter and position of the disc valve. When the valve is positioned at a distance from the seat where viscous effects are no longer important the "Bernoulli" effect begins to appear. This effect is most strongly exhibited when a large disc valve surface area is used. In this region the flow may separate or reattach to the valve seat downstream of the orifice port. This depends upon the pressure gradient between the entrance and exit of the valve, the diameter of the valve disc, and the Reynolds number at the entrance. At the point of reattachment a minimum pressure occurs. The point of location of the minimum pressure begins to shift toward the exit of the valve when the disc valve is displaced further from its seat. Flow separation occurs, and the lift force begins to approach the lift force of a jet impinging upon a flat plate. For extremely large disc valve positions, a loss of momentum occurs due to the flow around the valve.

In the development of the unsteady flow equations, Part I, provisions were made to include the reduction in flow areas at the entrance and exit of the disc valve. To achieve these two contraction coefficients CC_3 and CC_4 FIG (1), were introduced under the assumption that the flow pattern in the vicinity of the valve and orifice does not change appreciably from the static flow pattern. The exit contraction coefficient CC_4 was estimated from the free stream line data of an axisymmetric three-dimensional jet impinging upon a flat plate along with the experimental discharge coefficient data presented in REF (6). The contraction coefficient CC_3 at the entrance section of the disc valve was also established from the lift force coefficient data presented in REF (6). A set of coupling equations were developed using a momentum and continuity balance to relate both contraction coefficients for various classes of flow, REF (7).

COMPARISON OF UNSTEADY AND QUASI-STEADY MODELING TECHNIQUES

Disc valve displacement time histories comparing the quasi steady modeling technique with the unsteady modeling technique are presented in FIGS (2) and (3) as a function of crank angle position for a disc-valve to orifice diameter of 1.5. Two cases are considered, the first assumes initially ambient air in the piston chamber with air discharging through the disc valve system also into an ambient medium. The second case assumes a discharge to suction pressure ratio of 4:1 with the pressure initially in the piston chamber at an ambient state. Both disc valve displacements are considered from the instance when the reciprocating piston is in its bottom dead center position.

The mass flow equations presented in REF (1) were used along with the assumed discharge coefficient shown in FIG (7) as a function of the disc valve's static displacement for the quasi steady technique. The effective flow area was defined as $A_{ef} = \pi C C_4 d_p h_o$. The flow force acting on the disc valve as a function of its static displacement was determined from the static lift force coefficient data shown in FIG (4). It was also assumed that the momentum of the disc valve is completely eliminated upon impact with its seat.

With reference to FIG (2), the quasi steady modeling technique when compared with the unsteady modeling technique predicted a lower mean disc valve displacement with increased peak to peak oscillatory amplitudes. The displacement oscillations were more pronounced at the higher piston crank angle. The difference in the two time histories can be explained as follows. As the disc valve lifts from its seat there is a finite time before the flow in the disc valve restriction passage begins to develop, owing to the inertia of the gas. As a consequence higher pressures result in the piston cylinder causing the disc valve to displace further from its seat. This occured at approximately 45 degrees of crank angle rotation. The disc valve displacements predicted by the quasi steady theory are lower since the fluid velocity in the restriction passage responds instantaneously to an increase in cylinder pressure. At 45 degrees of crank rotation the mechanical spring force of the disc valve overrides the pressure force and the mechanical disc valve's inertia force. However, since the pressure force is increasing in the piston cylinder, the disc valve reverses its direction and continues to displace from its seat. During the motion the pressure will be greater than that predicted by the quasi steady modeling theory. The reverse situation occurs at approximately 110 degrees of crank rotation. Here, most of the fluid has been discharged from the reciprocating piston cylinder chamber, thus causing the cylinder pressure to decrease at a rate faster than the air can be compressed. The disc valve thus begins to move towards its seat. After a while the cylinder pressure will once again try to increase but not at a rate sufficient to overcome the inertia of the discharging fluid. The disc valve hence momentarily begins to move away from its seat, then towards it. At this point of cylinder crank angle rotation, 160°, the disc valve accelerates faster as it approaches the seat, since in this region the static lift force coefficient decreases at small static valve positions. Thus, the cylinder pressure increases and the disc valve executes its final oscillation before the valve closes. Again the disc valve oscillations are more pronounced using the quasi steady modeling technique for cylinder crank rotation angles greater than 140° since this modeling technique does not include the inertia of the fluid in the orifice and valve's passage.

Both valve displacement time histories compare quite favorably for a discharge to suction pressure ratio of 4:1. It is to be noted, however, that the peak displacement amplitude computed by the unsteady modeling technique is slightly higher than that computed by the quasi-steady modeling technique.

PARAMETRIC INVESTIGATION

Some computed results for the discharge disc valve system under study are presented in FIGS (5) through (11) for discharge to suction pressure ratios of 1:1. The effects of change in the mechanical disc valve spring rate on the displacement of the disc valve and pressures in the four control volumes are shown in FIGS (5) through (8) for mechanical spring loaded disc valve natural frequencies of 300 and 500 cps respectively. Disc valve displacement time histories are also shown in FIG (9) for the situation when the mass and spring rate of the spring loaded disc valve-mechanical spring system was assumed at twice that of the first system which assumed a natural frequency of the mechanical spring loaded disc valve of 300 cps as well as the effect of damping.

FIG (10) refers to the case where both mass and spring rate are ten times that of the standard system. The flow becomes supersonic at 65 degrees because of the large cylinder pressure built up. FIG (11) shows the behaviour of the valve when it has a very wide seat. The valve hovers close to the seat because of a pronounced Bernoulli effect. Supersonic velocity was reached at 110 degrees crank angle and the computation was terminated. It was found that as the seat width decreases, the time of hovering due to the Bernoulli effect decreases until hovering practically disappears from narrow seats. It is felt that this hovering phenomena accounts partially for what is labeled in literature the sticktion effect.

As expected, the response of the disc valve is affected by the choice of the mechanical systems natural frequency. During the half cycle of piston motion the frequency of oscillation of the disc vale changes and is dependent upon its mean amplitude of displacement. Unlike the case where the mechanical spring loaded disc valve natural frequency of 300 cps was assumed, the motion of the disc valve, for the case when a natural frequency of 500 cps was assumed, seems to be initially retarded over a greater portion of the piston's motion. There, the pressure, P_3 , between the disc valve and seat decreased below the discharge pressure and attained a minimum approximately at a piston crank angle rotation of 35 degrees.

Consider the response of the discharge disc valve system shown in FIG (5). The disc-valve begins to displace from its seat when the cylinder pressure increases above the discharge pressure. At very small displacements, viscous forces predominate in the disc valve's passage causing the pressure to increase due to compression of air in the piston cylinder. As the disc valve displaces further from its seat, the pressure between the disc valve and seat decreases and the motion of the disc valve retards temporarily. However since the pressure in the piston cylinder is increasing a sufficient force develops on the disc valve causing it to displace yet further. The disc valve continues to displace from the seat until the spring force of the mechanical spring of the disc valve counteracts the inertia force of the disc valve and pressure force acting on the disc valve. The disc valve then moves towards the seat and begins to oscillate about an increasing mean amplitude nearly coincident with the natural frequency of the mechanical disc valve system. For large disc valve mean displacement amplitudes, a pure jet begins to develop between the disc valve and seat. The pressures in the valve's restriction passage, P_2 and P_3 , thus began to communicate with the discharge exit pressure.

CONCLUSION

In an attempt to gain some understanding of the dynamic behavior of a compressor valve system and flow, a mathematical model was developed, in Part I, which considered the unsteady fluid dynamic modeling of a reciprocating compression discharge disc valve system. In general, it was found that the unsteady modeling approach did predict valve displacement time histories which differ from those computed by the quasi steady technique. For most instances, a higher mean amplitude of displacement with a reduced peak to peak oscillating amplitude was computed by the unsteady modeling technique. This type of valve displacement response can be explained by the inclusion of the inertia of the fluid in the disc valve's passage which is neglected when the quasi steady approach is used. The results shown in this paper were not verified experimentally. However, similar findings were also noted by other investigators when the quasi steady theory was compared with experimental data. The usualpprocedure in the quasi steady approach is then to induce empirically viscous clamping until good solution agreement is reached. This introduced damping is in some cases unreasonably high and affects other solution features. Thus, it is felt that it is of advantage to model some operating situations by unsteady flow. Especially if the tendency of the future is toward still higher compressor speeds. Unfortunatly, the technique presented in this paper leads to large computer time expenditures and it is, therefore, recommended that future work should be directed toward simplification of the unsteady approach.

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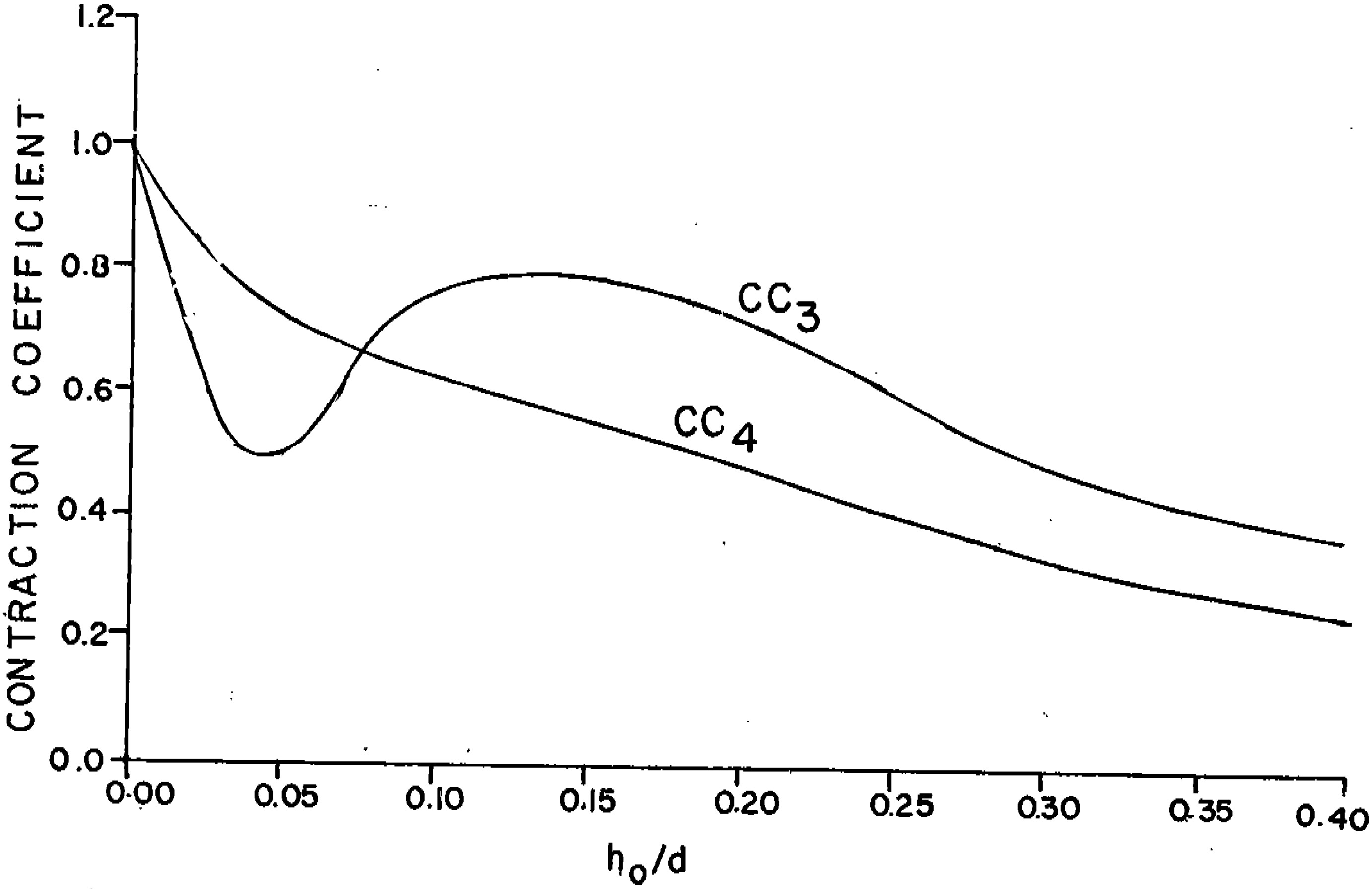


FIGURE 1 ASSUMED DISC VALVE ENTRANCE AND EXIT CONTRACTION COEFFICIENTS

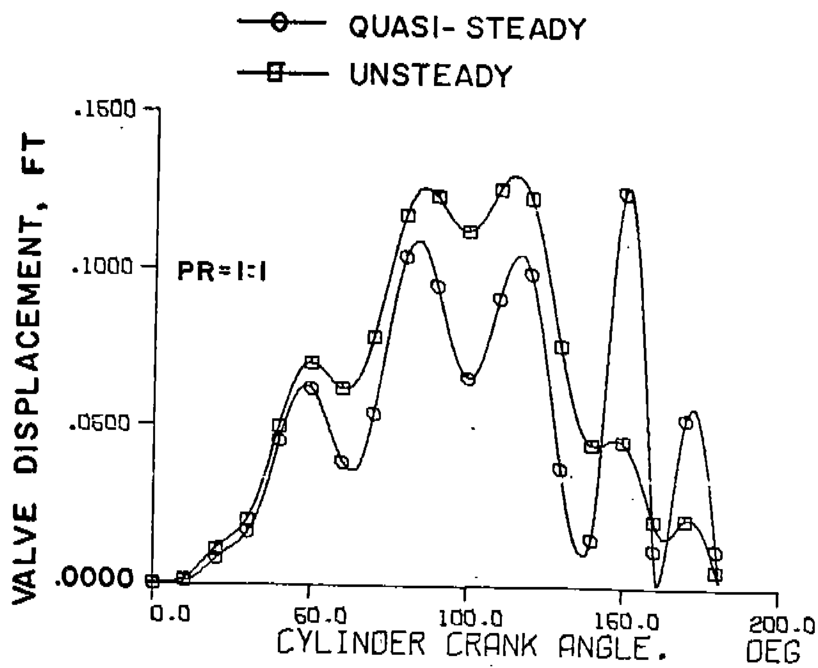


FIGURE 2

COMPARISON OF QUASI-STEADY AND UNSTEADY VALVE DISPLACEMENT, $\omega_n = 300$ cps

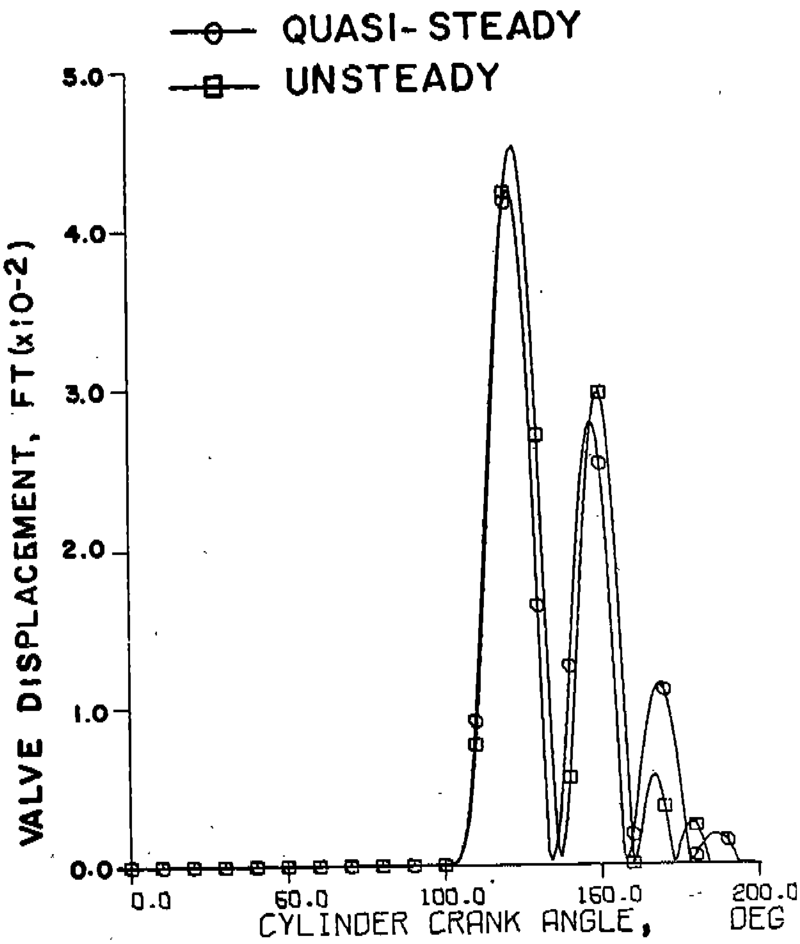


FIGURE 3

COMPARISON OF QUASI-STEADY AND UNSTEADY VALVE DISPLACEMENT, ω_n=300 cps, PR = 4:1

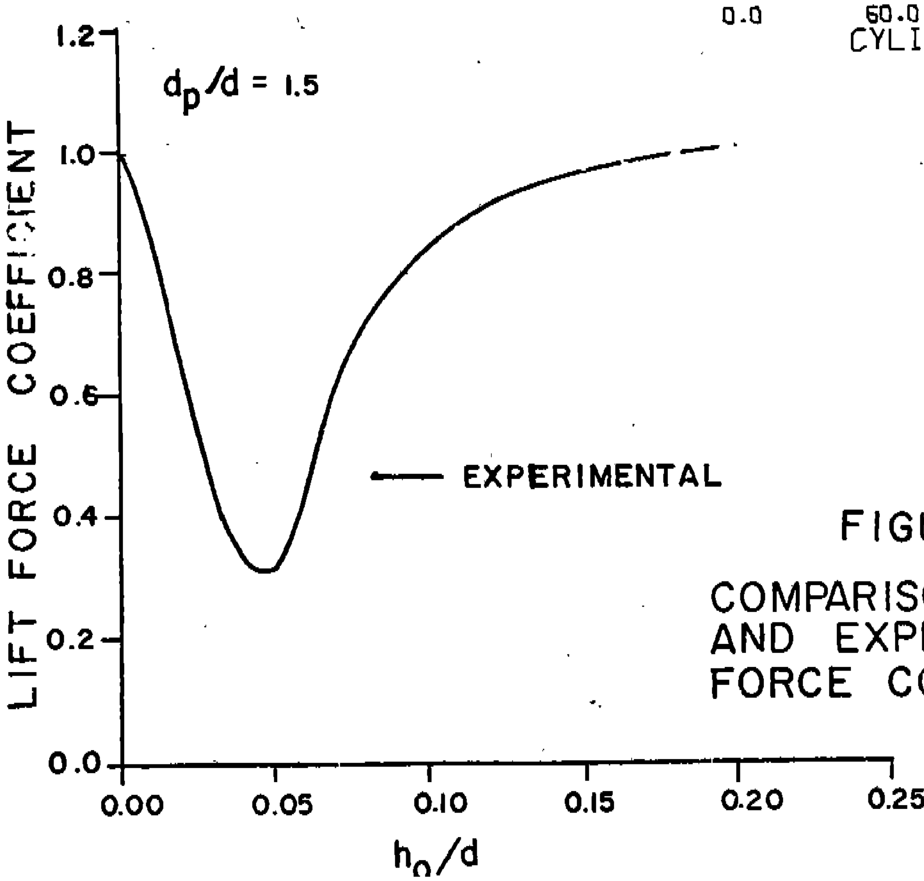


FIGURE 4

COMPARISON OF CALCULATED AND EXPERIMENTAL LIFT FORCE COEFFICENT

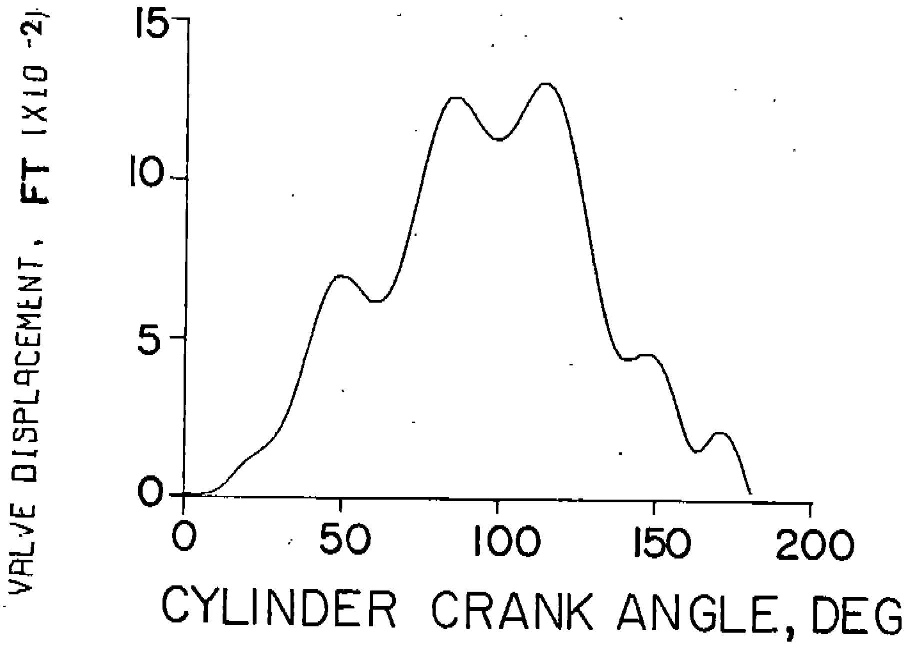


FIGURE 5

VALVE DISPLACEMENT
TIME HISTORY, ω_n= 300 cps

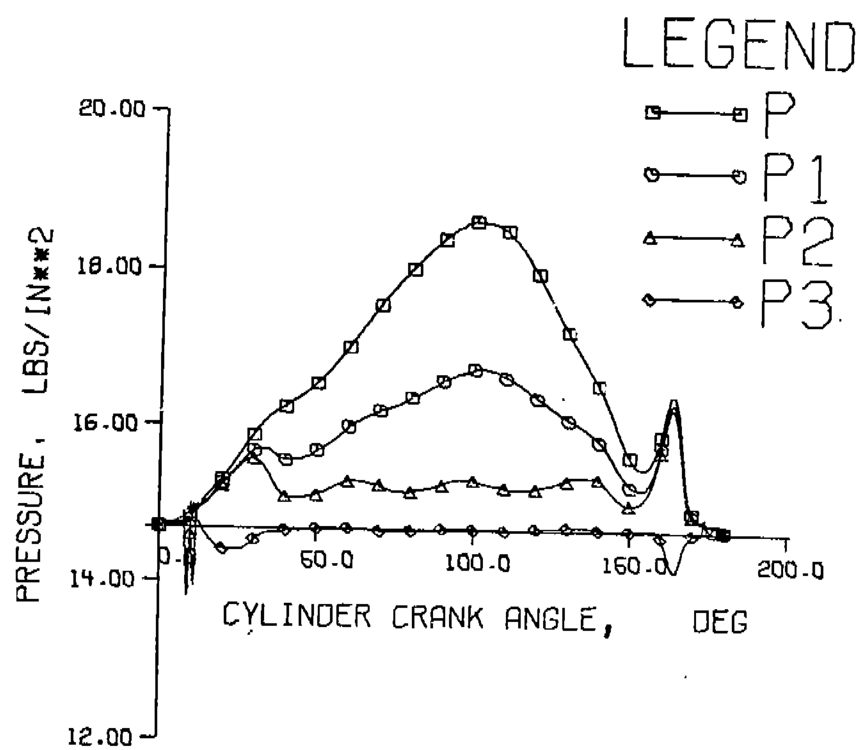


FIGURE 6

PRESSURE TIME
HISTORY, ω_n= 300 cps

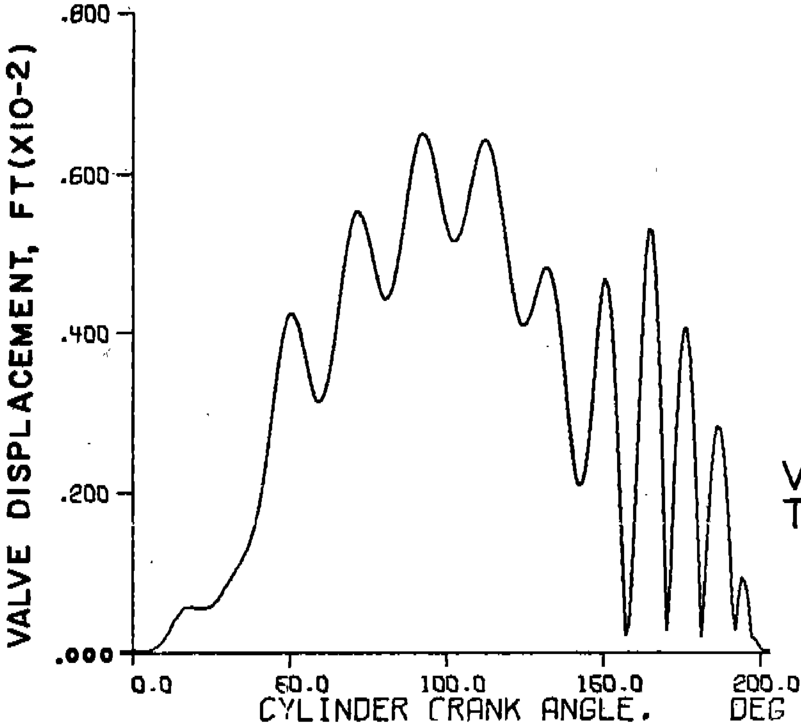
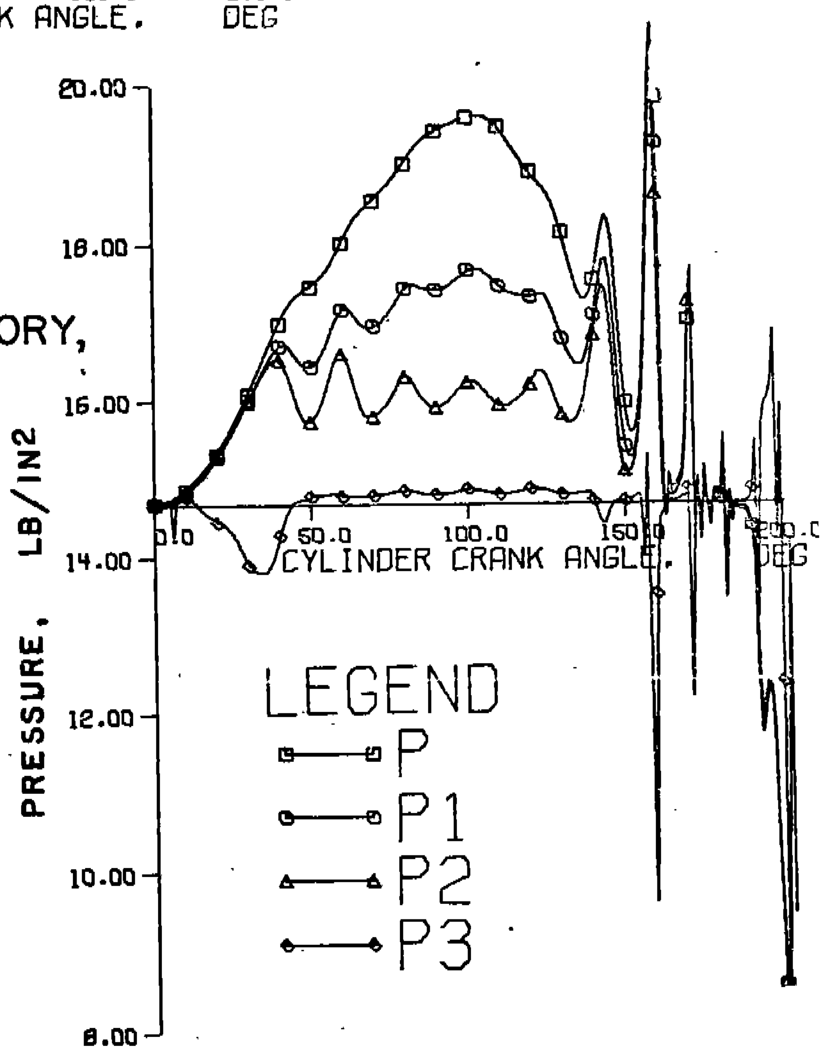


FIGURE 7

VALVE DISPLACEMENT TIME HISTORY. ω_n=500cps

FIGURE 8

PRESSURE TIME HISTORY, ω_n= 500 cps



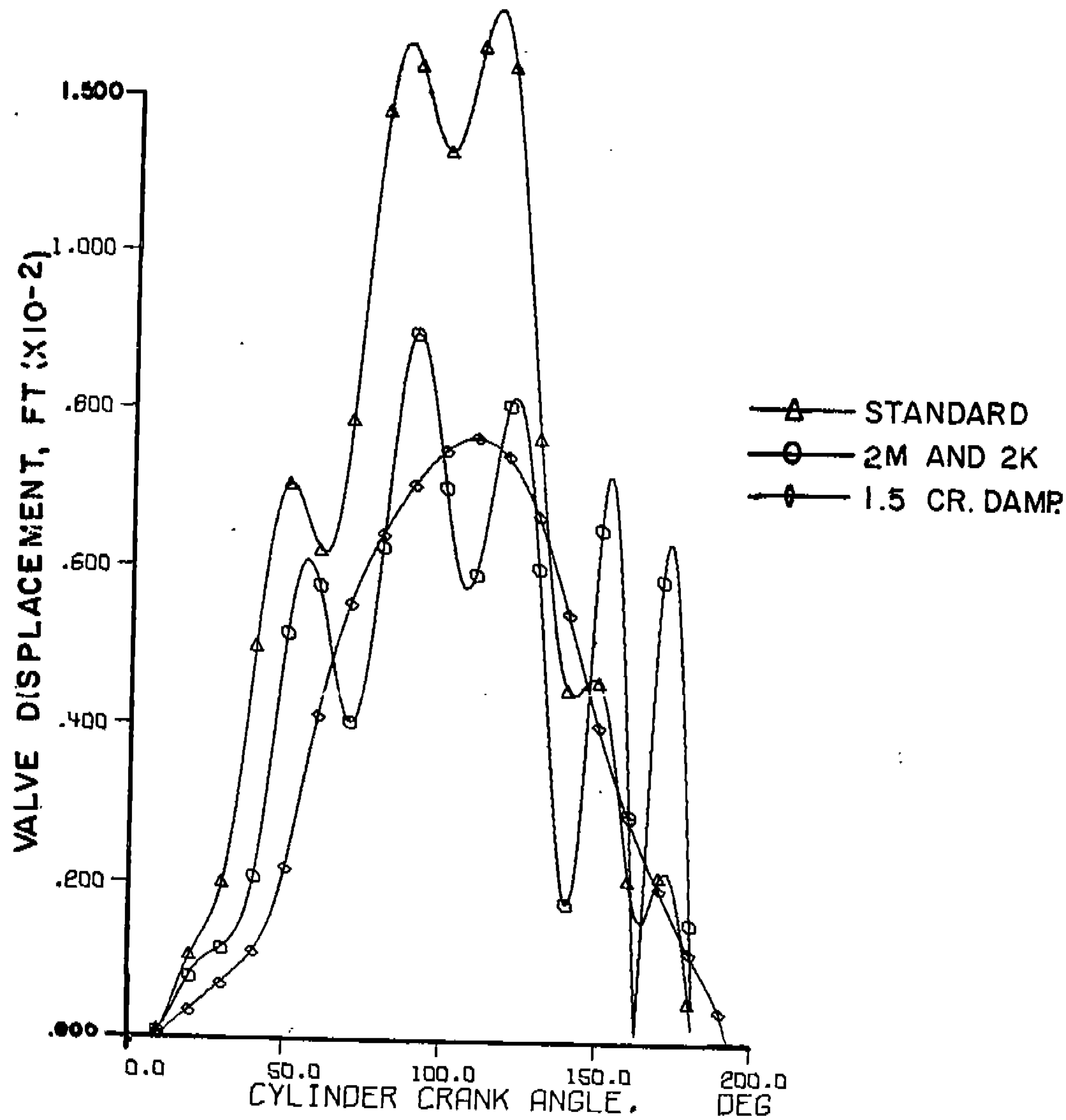


FIGURE 9 COMPARISON OF VALVE DISPLACEMENT TIME HISTORIES, ω_n=300cps

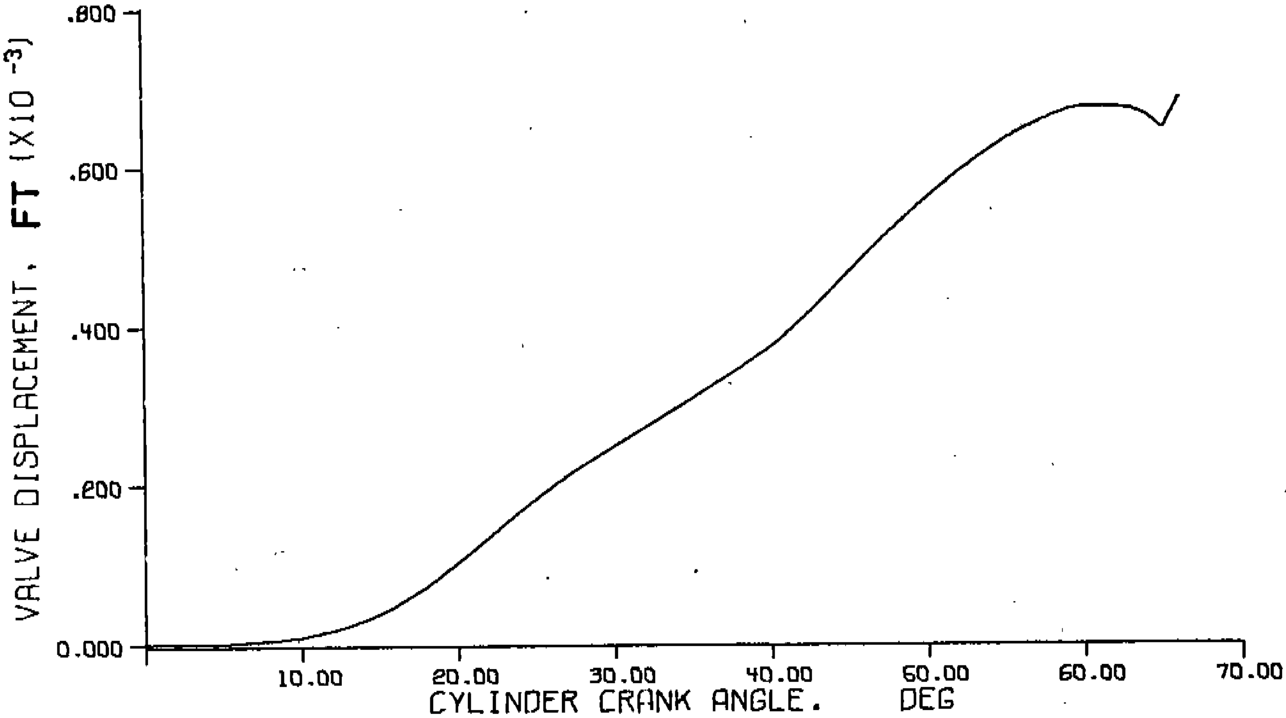


FIGURE 10 VALVE DISPLACEMENT TIME HISTORY, ω_n=300 cps, 10K AND 10M

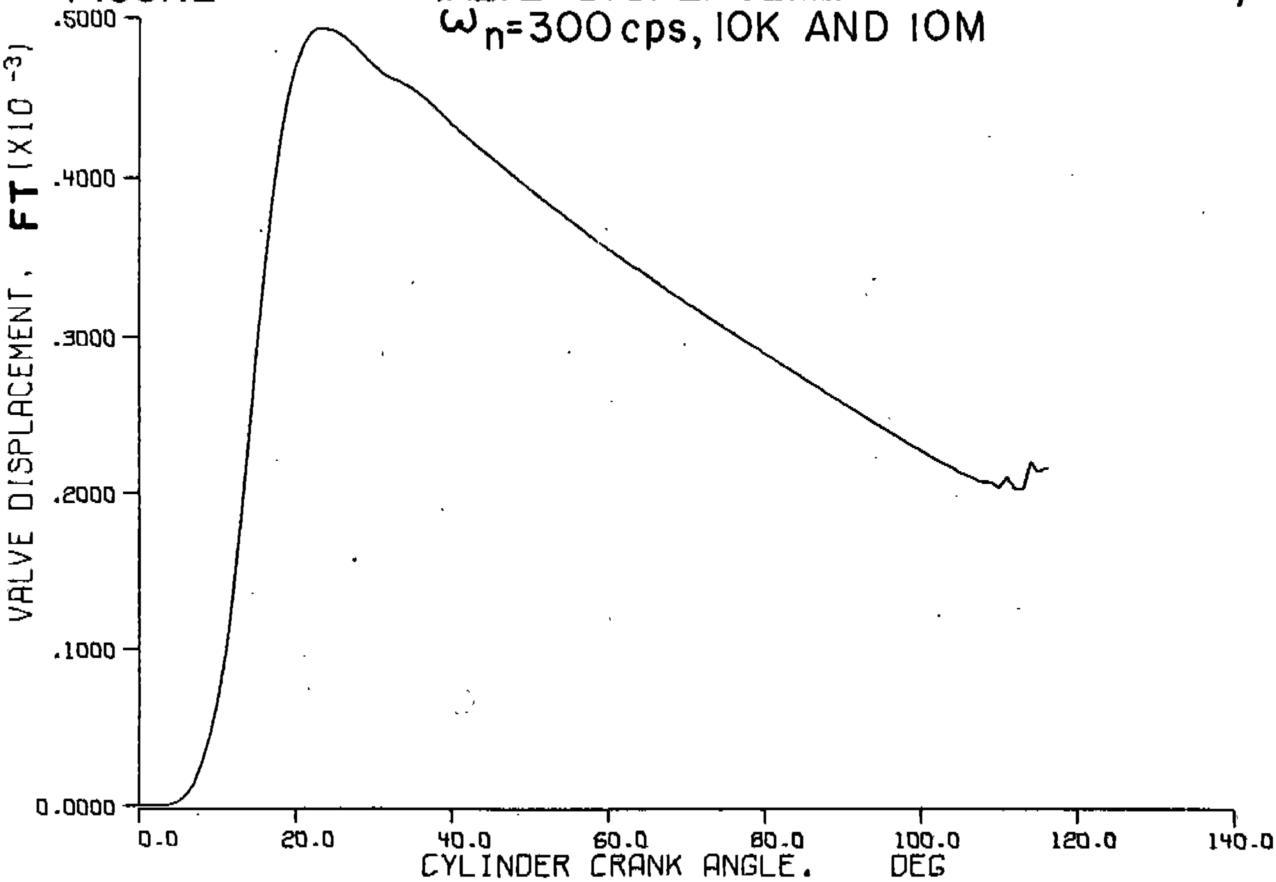


FIGURE 11 VALVE DISPLACEMENT TIME HISTORY, ω_n=300cps, d_p/d = 2.5